A SINGLE POINT COMPUTATION

FOR A

STEAM SPACE POWER PLANT

January 11, 1963

ASTRA, Inc.

Raleigh, North Carolina

\- - NOTE - -

The methods and results contained in this report are preliminary.

More refined radiator optimization methods and results are contained in ASTRA 205-1.2.3, "The Optimum Radiator for a Steam Space Power System." - X63-14520 However, the radiator weights given in this report are too high since meteoroid damage criteria have twice been revised (decreased) since the report was published.

H. R. Kroeger
10-29-64

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Copy No. 13

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FOR A
STEAM SPACE POWER PLANT

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TABLE OF CONTENTS

TABLES

ILLUSTRATIONS

I. INTRODUCTION

A. REASON FOR THIS REPORT

For some years ASTRA has had an interest in and has supported some analytical work with space power systems utilizing H_2O as a working fluid. In August, 1962, ASTRA undertook, for the NASA, to "provide a set of space radiator configurations and weights for various radiator parameters," for steam space power systems.

A preliminary computation of one point (30 ekw and T_R = 390 F) confirmed that small steam space power units were weight competitive with other system concepts. The computed system specific weight ($\sim$ 41 lb/ekw) was obtained for a concept utilizing conventional materials and temperatures. On this basis an early report of the radiator computation was requested in order to allow detailed checking of the assumptions, input data, and computational approach. This is that report.

B. TECHNICAL BACKGROUND

Effective space radiators are obviously of the tube-and-fin type. The tubes transport the heat to the fins and the fins conduct the heat to the point of radiation. The optimum radiator must take into account the transport fluid properties as well as the radiator material characteristics.

An optimum radiator (for any one temperature) is obtained by providing (1) an optimum fin configuration, (2) an optimum tube spacing, (3) an optimum pressure drop (temperature drop), and (4) an optimum header configuration. An optimum system also requires consideration of various radiator temperatures.

For the steam system the light metals aluminum and beryllium can be considered. Both have high thermal conductivities and, on a weight basis, provide excellent protection against meteoroid penetration.

Steam itself is an excellent working fluid. It transports about 6 - 7 times as much heat as mercury vapor and the ratio $\frac{dP}{dT}$ (for saturation conditions) is quite high. This allows the use of radiator tubes with an L/d ratio about 10 - 20 times that postulated for condensing metallic vapors.

C. SUMMARY RESULTS

1. Radiator Specifications (30 ekw)

a.	Type		- Tube-and-Fin
b.	p (probability of avoiding puncture for one year, Lewis Research Center approach)		- ~ 0.89
c.	Equivalent Space Temperature		- $440^{o}R$
d.	Material		- Aluminum
e.	ϵ (emissivity)		- 0.85
f.	α (absorptivity)		- 0.25
g.	Duty		- 220 tkw
h.	Inlet Temperature		- $400^{o}F$
i.	Inlet Pressure		- 248 psia
j.	Inlet Condition		- Saturated Vapor
k.	Outlet Temperature		- $382^{o}F$
l.	Outlet Pressure		- 200 psia
m.	Outlet Condition		- Saturated Liquid
n.	Number of Tubes (for parallel flow)	(N)	- 36
o.	Length of Tubes	(L_t)	- 40 ft
p.	Tube Spacing		- 0.606 ft (7.27 in)
q.	Tube I.D.	(d)	- 0.0105 ft (0.126 in)
r.	Tube Wall Thickness	(t_t)	- 0.020 ft (0.24 in)
s.	Tube Outside Diameter	(D)	- 0.0505 ft (0.606 in)
t.	Fin Thickness	(t_f)	- 0.0013 ft (0.0156 in)
u.	Fin Width	(B)	- 0.556 ft (6.672 in)
v.	Radiating Area		- 1760 ft^2
w.	Configuration (each section of two sections)		- 20 x 22 ft
x.	Radiator Weight (including headers and coolant)		- ~~680 lb~~ < 500 lb
y.	Specific Weight		- ~~3.09 lb/ekw~~

OBSOLETE. RADIATOR WEIGHT NOW LOWER DUE TO LATER METEOROID DATA

HRK 10-29-64

2. System Criteria

a.	Plant Size	- 30 ekw
b.	Working Fluid	- Water
c.	Type of Cycle	- Rankine
d.	Maximum Temperature	- 1200 F
e.	Minimum Temperature	- 360 F
f.	Maximum Pressure	- 1200 psia
g.	Minimum Pressure	- 200 psia
h.	Overall System Efficiency	- ~ 12%

3. System Weights

a.	Radiator	- 680
b.	Turbine-Pump-Generator	- 70
c.	Reactor	- 270*
d.	Recuperator	- 60
e.	Miscellaneous	- 140
f.	Total for System	- 1220
g.	Specific Weight	- 41 lb/ekw.

* Estimated weight for a once through direct steam generating reactor.

II. THE OPTIMUM FIN CONFIGURATION

For the tube-and-fin radiator it has been shown that there is an optimum (maximum heat radiated per pound of fin) relationship between the fin width and thickness.

A fin conduction parameter is defined as:

$$N_c = \frac{\sigma \epsilon T^3 B^2}{2kt_f}$$

where,

N_c = conduction parameter

σ = Stefan-Boltzman's constant = 0.174×10^{-8}

ϵ = emissivity of the fin = 0.85

T = temperature of the tube surface - $^{\circ}R$

B - fin width - ft (between tubes)

k = thermal conductivity of the fin material

t_f = fin thickness - ft.

Although k is a function of temperature, and the fin temperature decreases with distance from the tube, for this analysis k will be evaluated at the tube surface temperature. This is slightly conservative.

The relationship between t_f and B is then:

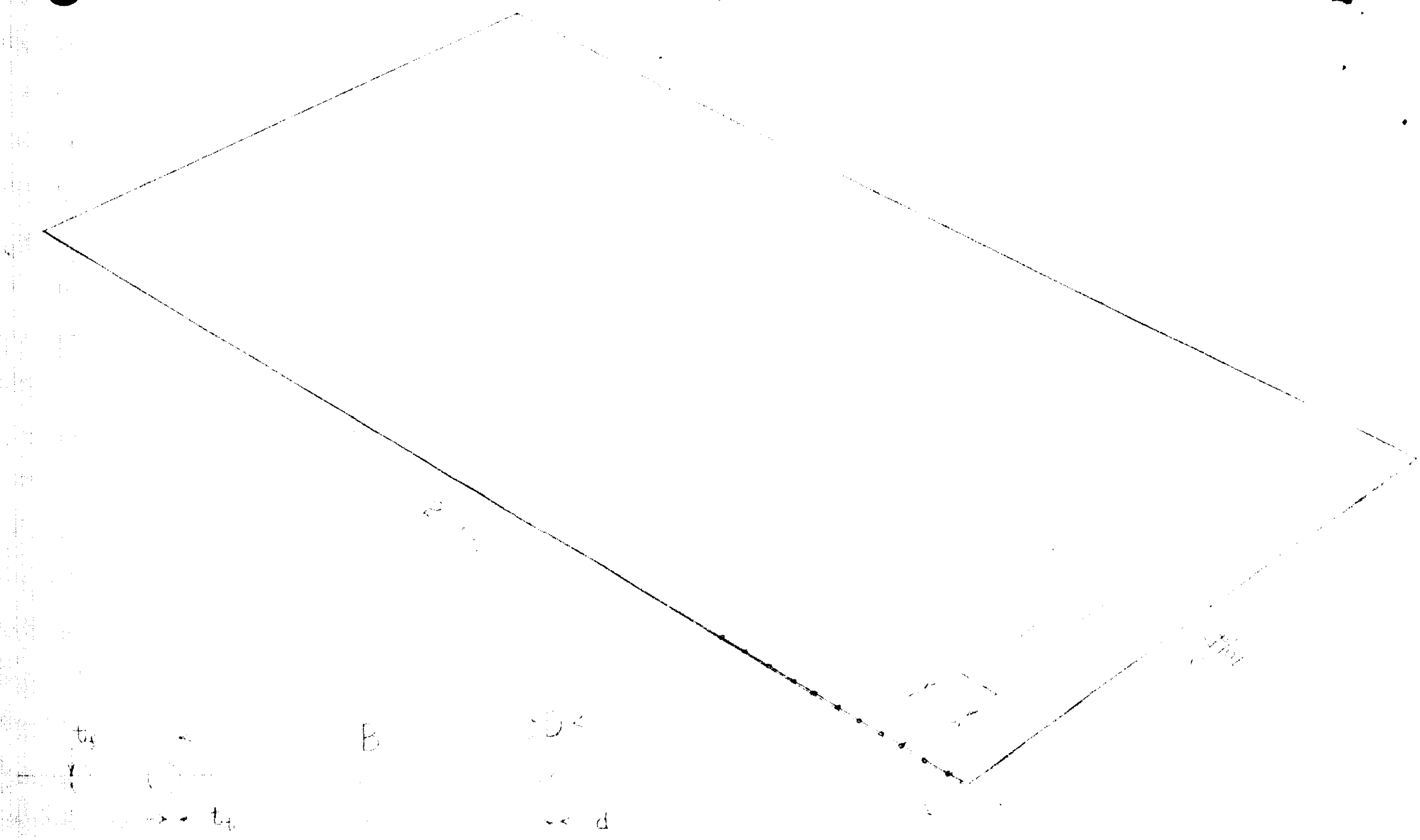
$$t_f = \frac{0.0739 \times 10^{-8}\ T^3}{k\ N_c}\ B^2 = C_2 B^2$$

where

$$C_2 = \left(\frac{0.0739 \times 10^{-8}\ T^3}{k\ N_c} \right) .$$

The value of N_c which gives an optimum fin configuration depends independently on the ratio (θ_s) of effective space temperature to radiator temperature and the radiator configuration ratio B/D (see Figure 1).

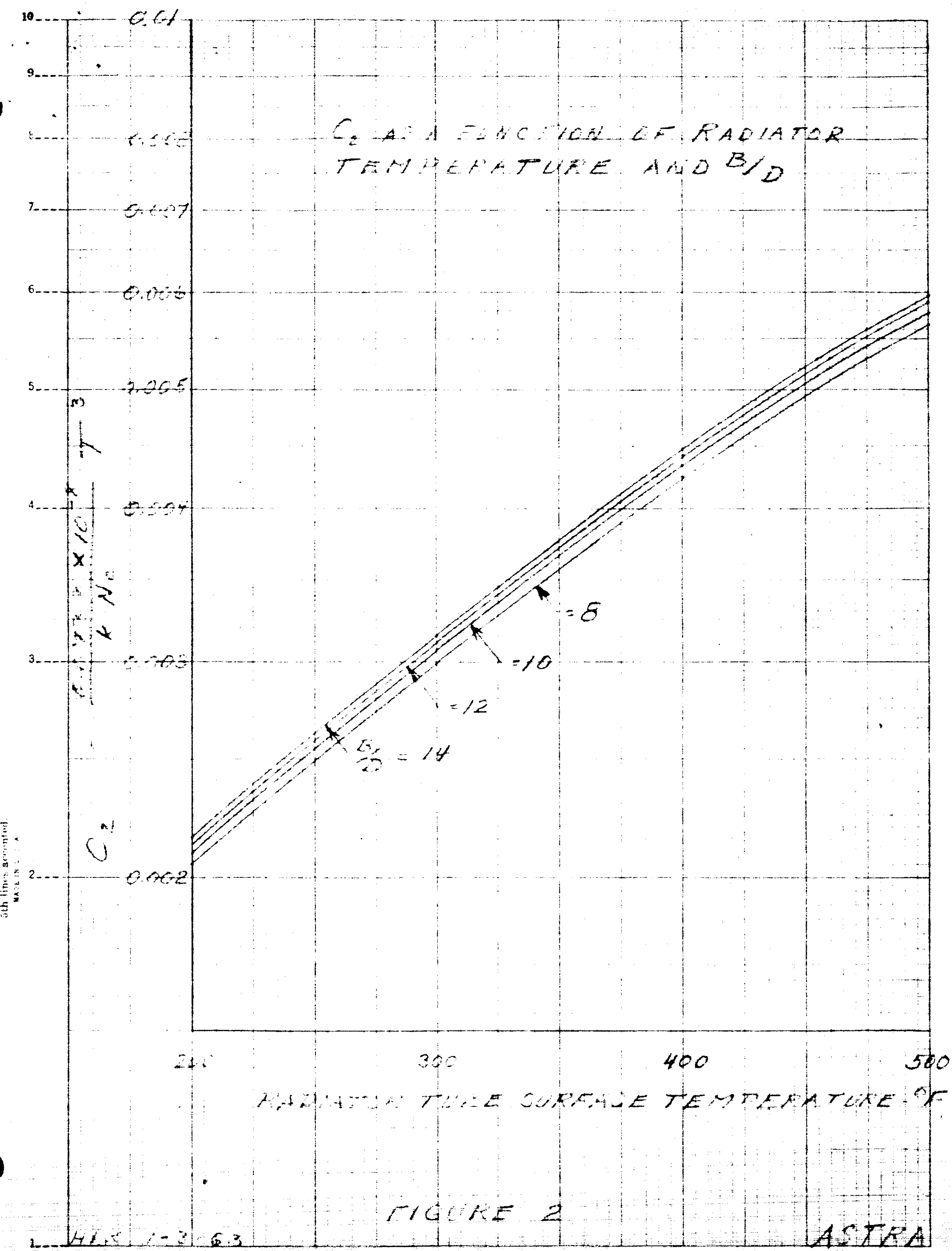
Figure 2 evaluates C_2 for various temperatures and values of optimum N_c.



Section A-A

A SCHEMATIC OF ONE SECTION OF A SPACE STEAM CONDENSER

FIGURE 1



C_2 AS A FUNCTION OF RADIATOR TEMPERATURE AND B/D
0.01
0.008
0.007
0.006
0.005
0.004
0.003
0.002
B/D = 14
= 12
= 10
= 8
200
300
400
500
RADIATION TUBE SURFACE TEMPERATURE °F
ASTRA

FIGURE 2

III. THE OPTIMUM TUBE SPACING

1. Condenser Duty

For the radiator configuration shown in Figure 1 and for a condensing heat transfer fluid the total heat radiated is:

$$Q_c = Q_t + Q_f$$

where Q_c = heat radiated from condenser,

Q_t = heat radiated from tubes only,

and Q_f = heat radiated from fins only.

For a single lone tube at a uniform temperature

$$\frac{Q_t}{N} = q_t \pi D L_t$$

where q_t = specific tube radiation in tkw/ft^2-hr

$$\left(q_t = \frac{\sigma \epsilon}{3413} (T^4 - T_S^4) \right)$$

N = number of tubes in the condenser

D = outside diameter of tube - ft

and L_t = length of a single tube - ft

T_S = space temperature, ^{o}R.

Similarly

$$\frac{Q_f}{N} = 2(q_f B L_t)$$

where q_f = specific fin radiation, $\frac{\text{tkw}}{\text{ft}^2\text{-hr}}$.

Since the average fin temperature is less than the tube temperature, $q_f < q_t$. It is convenient however, to relate these two factors. Thus, a fin effectiveness factor (η_o) is defined as:

$$\eta_o = \frac{q_f}{q_t}$$

and $\frac{Q_f}{N} = 2q_t\eta_o BL_t$.

The above expressions are valid for a flat plate radiator (i.e., where $D = t_f$). For an actual radiator it is necessary to introduce a shadow factor (angle factor, view factor, geometry factor) which is primarily a function of B/D. Thus, with these factors included, the total heat radiated from a single tube and fin is:

$$\frac{Q_c}{N} = q_t \pi D L_t F_t + 2q_t \eta_o B L_t F_f$$

or

$$\frac{Q_c}{N} = q_t L_t (\pi D F_t + 2\eta_o B F_f)$$

where F_t = shadow factor for the tube ($\sim$ 0.84 for steam radiators)

and F_f = shadow factor for the fins ($\sim$ 0.96 for steam radiators).

The total projected radiating area of a single tube and fin is:

$$\frac{A_c}{N} = 2L(D + B)$$

and the heat radiated per unit projected radiating area for the whole condenser is:

$$q_c = \frac{Q_c}{A_c} = \frac{q_t}{2}\left(\frac{\pi DF_t + 2\eta_o BF_f}{D + B}\right) .$$

2. Condenser Weight

The weight of the condenser (without headers) is just the sum of the tube and fin weights.

$$W_c = W_t + W_f.$$

The weight of a single tube and fin is

$$\frac{W_c}{N} = \frac{\pi}{4}\left(D^2 = d^2\right) \rho_t L_t + t_f B \rho_f L_t$$

where d = inside tube diameter - in.

ρ_t = density of tube material - lb/ft^3

ρ_f = density of fin material - lb/ft^3.

The condenser weight per unit projected area is:

$$w_c = \frac{W_c}{A_c} = \frac{\frac{\pi}{4}(D^2 - d^2)\rho_t + t_f B \rho_f}{2(D + B)} \quad .$$

For the case where the radiator tubes and fins are of the same material the total radiator weight per unit of heat rejected is:

$$\frac{W_c}{Q_c} = \frac{w_c}{q_c} = \frac{\rho_c\left(\frac{\pi}{4}(D^2 - d^2) + t_f B\right)}{q_t(\pi D F_t + 2\eta B F_f)}$$

where ρ_c = density of condenser material, lb/ft^3.

But $D = d + 2t_t$

where t_t = tube wall thickness - ft

$$D^2 = d^2 + 4dt_t + 4t_t^2$$

and $(D^2 - d^2) = 4t_t(d + t_t)$.

Also for the optimum (minimum weight fin) t_f and B are related by:

$$t_f = \frac{\sigma \epsilon T^3}{2kN_c} B^2$$

where the optimum value of N_c is used (see Figure 3). The thermal conductivities of the condenser materials are shown in Figure 4.

And

$$\frac{w_c}{q_c} = \frac{\rho_c}{q_t} \left(\frac{\pi t_t(d + t_t) + \frac{\sigma \epsilon T^3}{2kN_c} B^3}{\pi(d + 2t_t)F_t + 2\eta_o BF_f} \right) .$$

For ease in manipulation the above expression can be written:

$$\frac{w_c}{q_c} = \frac{\rho_c}{q_t} \left(\frac{C_1 + C_2B^3}{C_3 + C_4B} \right)$$

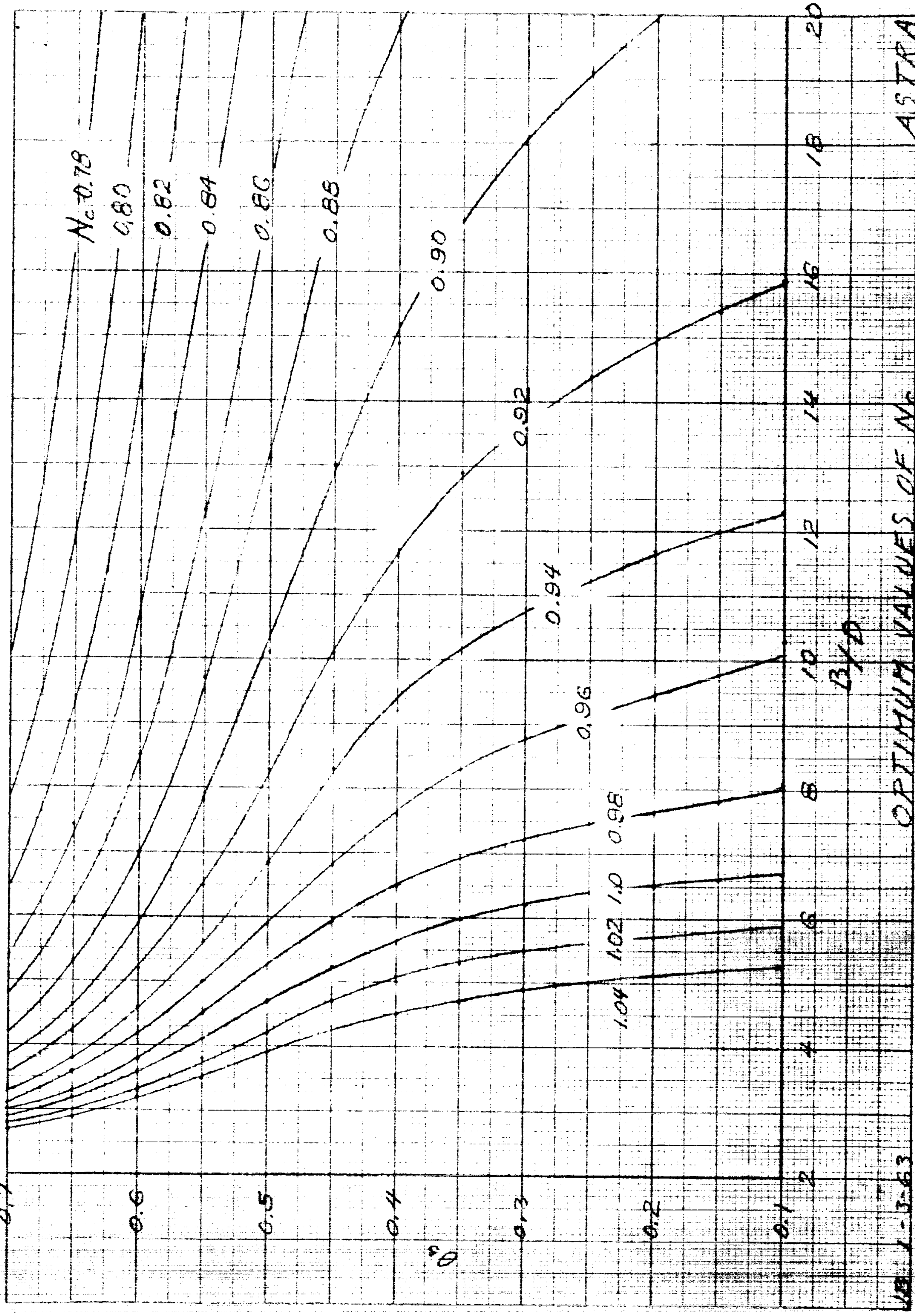
where $C_1 = \pi t_t(d + t_t)$

$$C_2 = \frac{\sigma \epsilon T^3}{2kN_c} = \frac{0.0739 \times 10^{-8} T^3}{kN_c}$$

$$C_3 = \pi(d + 2t_t)F_t$$

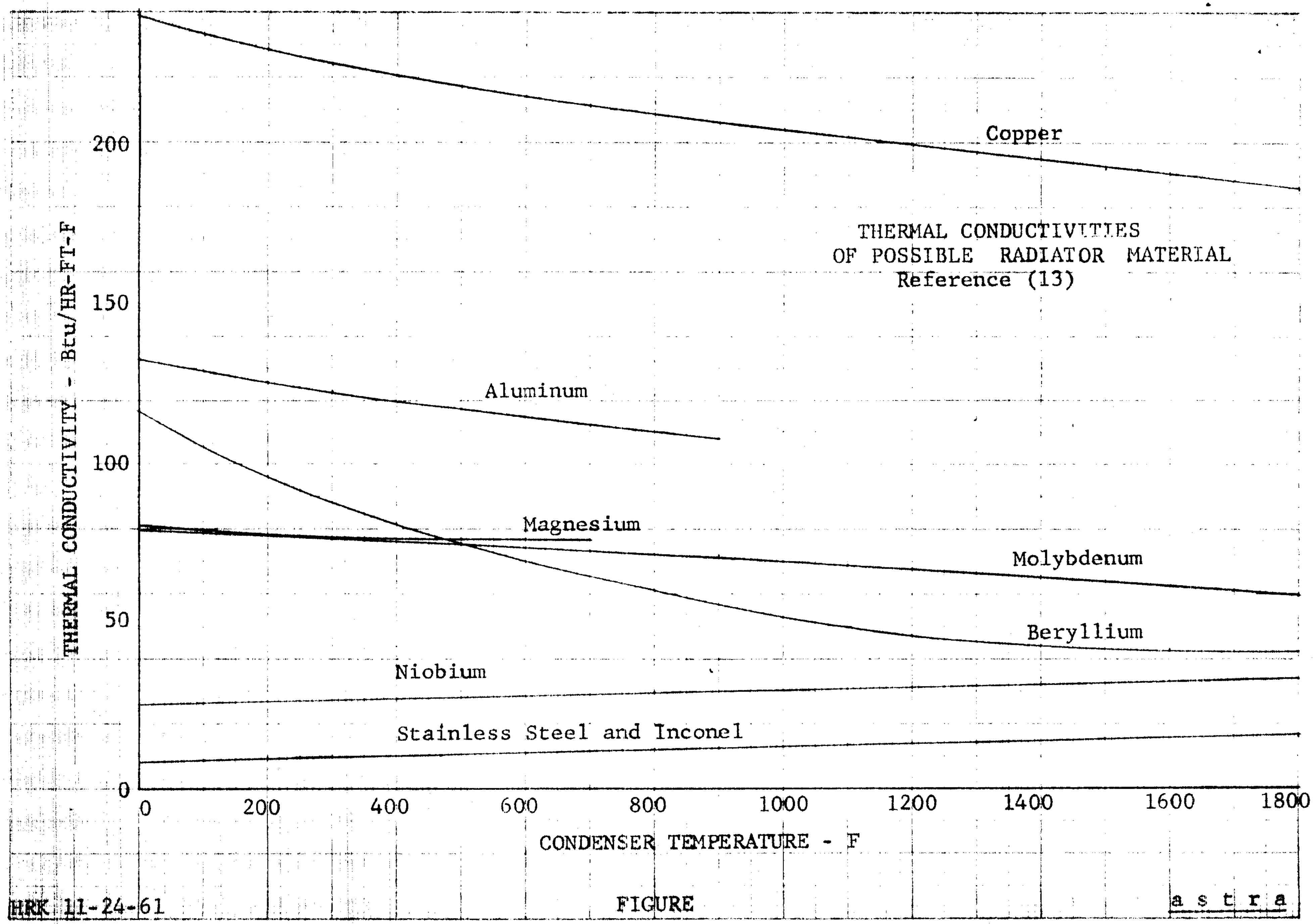
and $C_4 = 2\eta_o F_f$.

The value of B which gives an optimum value of the ratio $\frac{w_c}{q_c}$ can be determined from a plot of B vs $\left(\frac{C_1 + C_2B^3}{C_3 + C_4B} \right) = R$ for various assumed values of d, t_t, material, and tube radiating temperature. In order to reduce computation time the radiating temperature and tube wall thickness was assumed constant ($T = 390^{\circ}F$ and $t_t = 0.02$ ft). The material considered initially is aluminum.



N_c = 0.78
0.80
0.82
0.84
0.86
0.88
0.90
0.92
0.94
0.96
0.98
1.0
1.02
1.04
0.7
0.6
0.5
0.4
0.3
0.2
0.1
ϕ_0
2
4
6
8
10
12
14
16
18
20
B/D
OPTIMUM VALUES OF N_c
ASTRA
1-3-63

FIGURE 3


Copper
THERMAL CONDUCTIVITIES
OF POSSIBLE RADIATOR MATERIAL
Reference (13)
Aluminum
Magnesium
Molybdenum
Beryllium
Niobium
Stainless Steel and Inconel
THERMAL CONDUCTIVITY - Btu/HR-FT-F
200
150
100
50
0
0
200
400
600
800
1000
1200
1400
1600
1800
CONDENSER TEMPERATURE - F
HRK 11-24-61
FIGURE
a s t r a

Thus $C_1 = 0.02\pi(d + 0.02)$

$$C_2 = \frac{0.0739 \times 10^{-8}}{120 \times N_c} (390 + 460)^3$$

N_c from Figure 3

$$\theta_s = \frac{440}{850} = 0.518$$

k from Figure 4 (= 120)

$$C_3 = 0.84\pi(d + 0.04)$$

$$C_4 = 1.92\eta_o$$

η_o from Figure 5 ($\theta_s = 0.518$).

Table I details the computations d = 0.010 and four values of B/D. Figure 6 is a plot of these points (and others) and shows the values of fin width for minimum radiator weight.

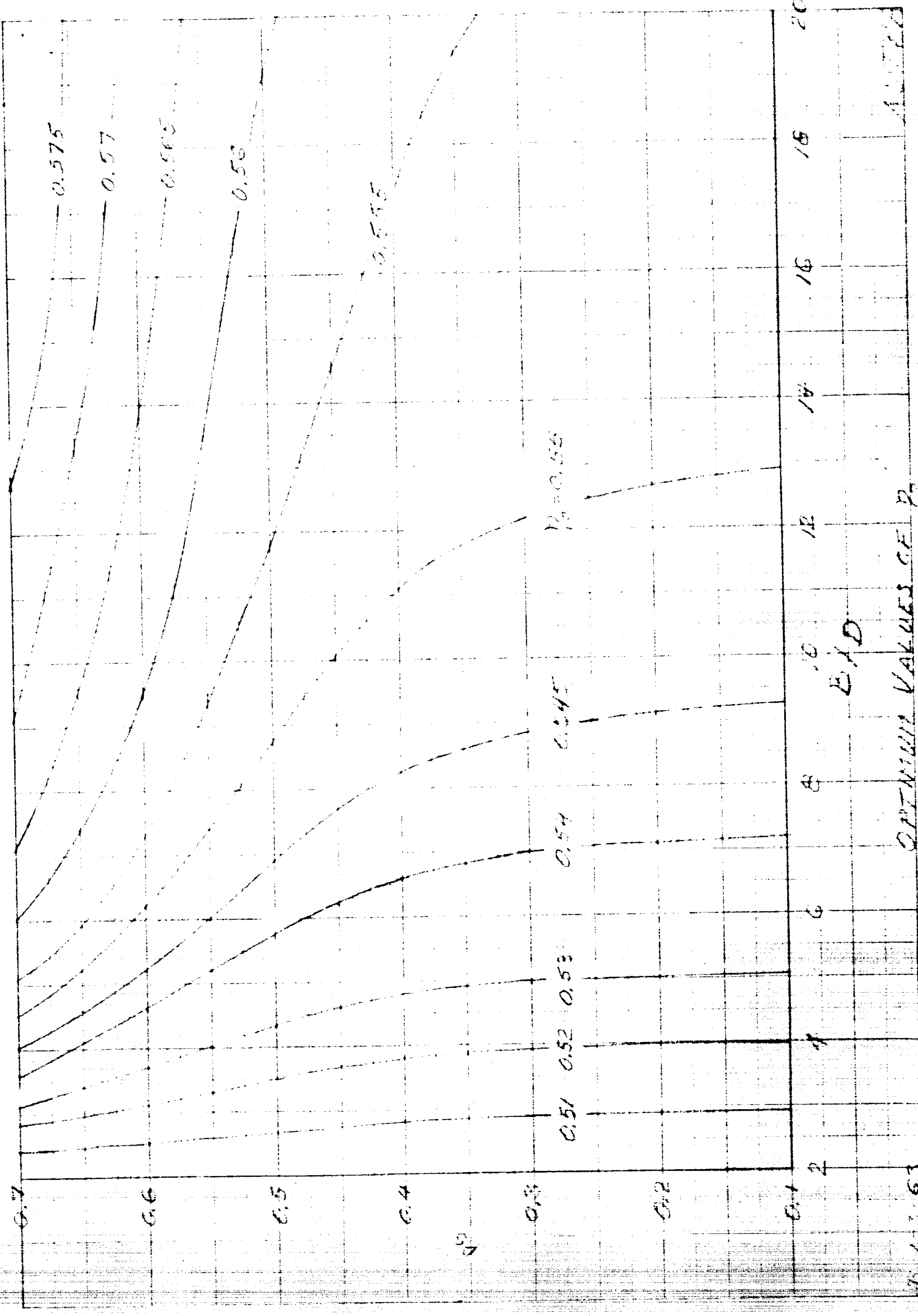

0.575
0.57
0.565
0.56
0.555
P_2 = 0.55
0.545
0.54
0.53
0.52
0.51
0.7
0.6
0.5
0.4
0.3
0.2
0.1
2
4
6
8
10
12
14
16
18
20
E/D
OPTIMUM VALUES OF P_2

TABLE I. THE WEIGHT FUNCTION R AS A FUNCTION OF FIN WIDTH

(d = 0.01 ft)

B/D	8	10	12	14
D	0.05	0.05	0.05	0.05
B	0.4	0.5	0.6	0.7
B^3	0.064	0.125	0.216	0.343
C_1	0.00188	0.00188	0.00188	0.00188
C_2	0.00409	0.00419	0.00427	0.00433
C_3	0.1319	0.1319	0.1319	0.1319
C_4	1.055	1.062	1.067	1.071
C_2B^3	0.000262	0.000525	0.000920	0.001482
C_4B	0.421	0.530	0.642	0.750
$C_1+C_2B^3$	0.00214	0.00241	0.00280	0.00336
C_3+C_4B	0.553	0.662	0.774	0.882
R	0.00387	0.00364	0.00362	0.00382

NOTE: Above based on $T_s = 440^oR$, $T = 390^oF$, $t_t = 0.02$ ft, and an all aluminum structure.

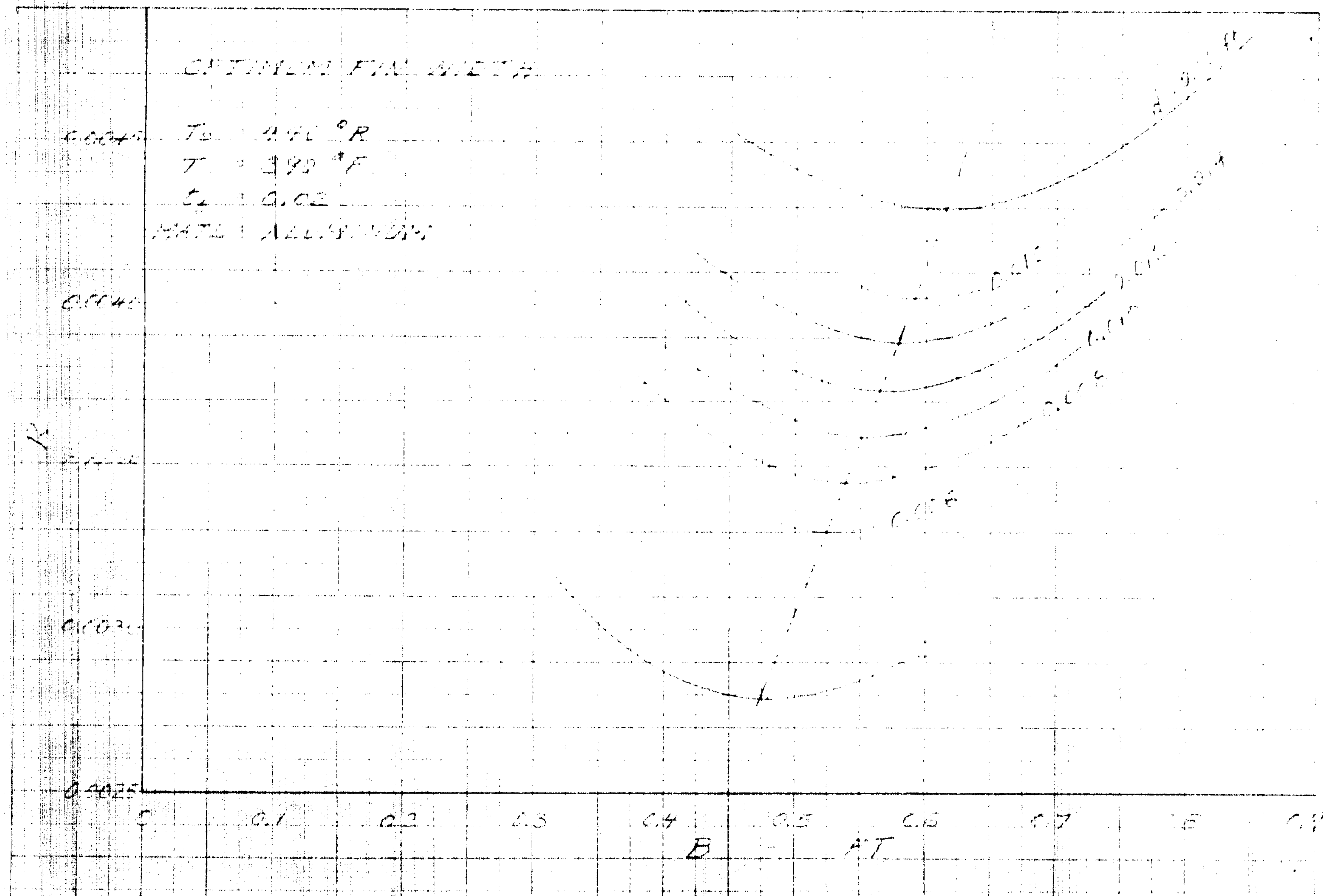


FIGURE 5

IV. THE OPTIMUM TEMPERATURE DROP ACROSS THE CONDENSER

Consider a single tube section of a tube-and-fin radiator (Figure 1). It has been shown (Section III) that the total heat radiated from the section is:

$$\frac{Q_c}{N} = \bar{q}_t L_t (C_3 + C_4 B)$$

where $\bar{q}_t$ = an arbitrary average value defined as the unit heat radiated (tkw/ft^2-hr) at a temperature just halfway between the inlet and exit temperatures.

It is assumed that working fluid enters the radiator as saturated steam and leaves as saturated water. For equilibrium conditions the heat radiated from a single tube and fin must just equal the total heat of condensation of the steam handled by the tube, i.e.,

$$\frac{Q_c}{N} = m\bar{h}$$

where m = the flow per tube in lb/hr,

and $\bar{h}$ = the heat given up by each pound of steam condensing in tkw/lb (for consistency with $\bar{q}_t$ units).

The flow (m) is related to the steam properties and the tube dimensions by the pressure drop relationship

$$\Delta P = 1.35 \times 10^{-13} \frac{m^2 v}{d^4} \cdot \frac{L_t}{d}$$

where ΔP = pressure drop across the condenser in psi,

and v = specific volume of the steam at the condenser inlet.

This pressure drop expression suggested by Kern is based on empirical results for condensing steam (see Appendix VI-B).

The results given by the expression are within the limits of accuracy given for more sophisticated (and complicated) expressions and are probably conservative.

Solving the above expression for the flow:

$$m = \left(\frac{d}{L_t} \cdot \frac{d^4 \Delta P}{1.35 \times 10^{-13} v} \right)^{1/2}$$

$$\text{and } m\bar{h} = d^2 \bar{h} \left(\frac{d}{L_t} \cdot \frac{\Delta P}{1.35 \times 10^{-13} v} \right)^{1/2} = Q_c/N.$$

Thus

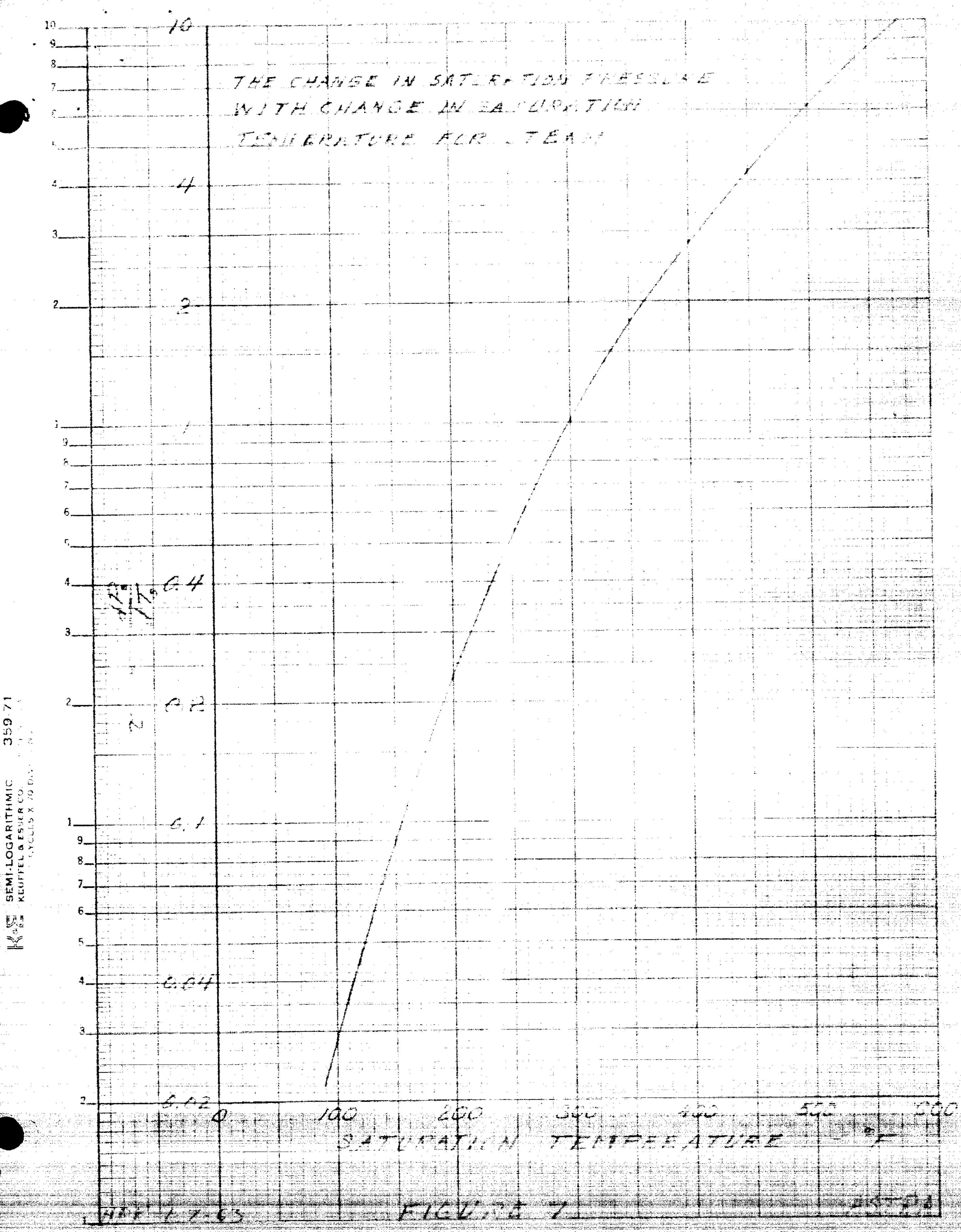
$$d^2\bar{h}\left(\frac{d}{L_t}\cdot\frac{\Delta P}{1.35\times 10^{-13}v}\right)^{1/2} = \bar{q}_t L_t (C_3 + C_4 B).$$

Solving for the pressure drop

$$\Delta P = \left(\frac{\bar{q}_t L_t}{d^2\bar{h}}(C_3 + C_4 B)\right)^2 \frac{L_t}{d}\cdot 1.35\times 10^{-13} v.$$

In the above expression $\bar{q}_t$, $\bar{h}$, v, and C_4 are all functions of temperature.

The change in the saturation temperature with a change in pressure relates ΔT and ΔP. If we define this relationship as $\tau = \frac{\Delta P}{\Delta T}$ then $\Delta P = \tau \Delta T$ (see Figure 7). The value of this τ is one of the primary factors that allows low temperature, steam systems to favorably compete with liquid metal systems. τ for steam is many times the τ for metal vapors and thus a considerably larger pressure drop (ΔP) may be used across steam radiators without incurring a large temperature drop (ΔT). Thus the optimum pressure drop for steam radiators is large compared to liquid metal radiators and therefore smaller tubes can be used, resulting in lighter weight.


THE CHANGE IN SATURATION PRESSURE
WITH CHANGE IN SATURATION
TEMPERATURE FOR STEAM
10
4
2
1
0.4
0.2
0.1
0.04
0.02
0
100
200
300
400
500
600
SATURATION TEMPERATURE °F
SEMI-LOGARITHMIC
KEUFFEL & ESSER CO.
359-71
APR 17 63
FIGURE 7

C_4 was assumed independent of ΔT to save computational labor. Moving other terms sensitive to ΔT to the left side of the equation results in:

$$\tau \Delta T \left(\frac{\bar{h}}{\bar{q}_t} \right)^2 = \left(\frac{C_3 + C_4 B}{d} \right)^2 \cdot \left(\frac{L_t}{d} \right)^3 \quad 1.35 \times 10^{-13} \times v.$$

Values of τ, $\bar{h}$, and $\bar{q}_t$ are determined for any condenser inlet temperature (T_i) and any ΔT. Thus a plot of $\tau \Delta T \left(\frac{\bar{h}}{\bar{q}_t} \right)^2$ (Table II and Figure 8) will allow the determination of ΔT if the right side of the above expression is evaluated.

For an assumed length of tube the temperature drop (ΔT) may be found for each of several tube diameters (d). Since $R_{(OPT)}$ has been determined (Figure 6) the specific weight can be computed from

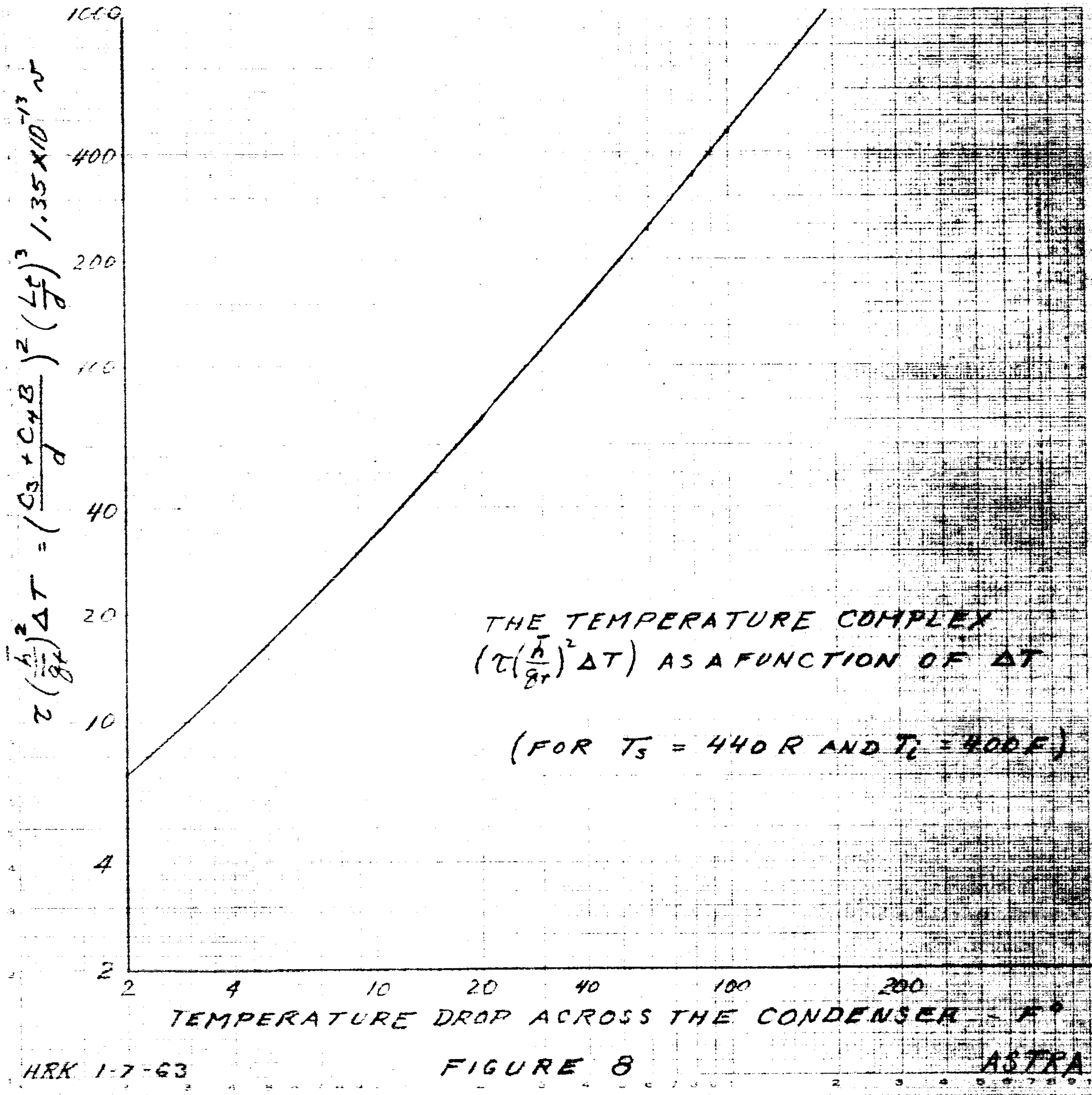
$$\frac{w_c}{q_c} = R_{(OPT)} \frac{\rho}{\bar{q}_t} \quad .$$

Table III details the computations for L = 60 ft for several values of tube diameter (d). Figure 9 shows the reduction in $\bar{q}_t$ for an increasing ΔT for a condenser with T_i = 400 F.

TABLE II. THE TEMPERATURE COMPLEX $\left(\tau \left(\frac{\overline{h}}{\overline{q}_t}\right)^2 \Delta T\right)$ AS A FUNCTION OF ΔT

(For $T_s = 440^\circ R$ and $T_i = 400^\circ F$)

ΔT	10	20	40	60	80	100
$\overline{T}$	395	390	380	370	360	350
T_e	390	380	360	340	320	300
τ	2.69	2.55	2.33	2.13	1.95	1.78
h_s	1201	1201	1201	1201	1201	1201
h_l	364	353	332	311	290	270
$\overline{h}$	0.2452	0.2483	0.2545	0.2607	0.2668	0.2726
$\overline{q}_t$	0.2150	0.2095	0.1989	0.1890	0.1793	0.1700
$\left(\frac{\overline{h}}{\overline{q}_t}\right)^2$	1.30	1.40	1.64	1.90	2.22	2.57
$\tau\left(\frac{\overline{h}}{\overline{q}_t}\right)^2 \Delta T$	34.9	71.4	152.8	242.8	346	457



1000
400
200
100
40
20
10
4
2
2
4
10
20
40
100
200
$\tau\left(\frac{\bar{h}}{g_r}\right)^2 \Delta T = \left(\frac{C_3 + C_4 B}{d}\right)^2 \left(\frac{L_t}{d}\right)^3 1.35 \times 10^{-13} \nu$
THE TEMPERATURE COMPLEX
$\left(\tau\left(\frac{\bar{h}}{g_r}\right)^2 \Delta T\right)$ AS A FUNCTION OF ΔT
(FOR T_s = 440 R AND T_l = 400 F)
TEMPERATURE DROP ACROSS THE CONDENSER - F°
HRK 1-7-63
ASTRA

FIGURE 8

TABLE III. TEMPERATURE DROP ACROSS THE CONDENSER FOR SEVERAL TUBE DIAMETERS

(L_t = 60 ft)

d	0.011	0.012	0.013	0.014
D	0.051	0.052	0.053	0.054
$B_{(OPT)}$	0.558	0.565	0.571	0.578
B/D	10.93	10.86	10.77	10.69
(C_3+C_4B)	0.728	0.738	0.748	0.758
$\left(\frac{C_3+C_4B}{d}\right)$	66.2	61.5	57.5	54.2
$\left(\frac{C_3+C_4B}{d}\right)^2 \times 10^{-4}$	0.451	0.378	0.331	0.294
$\left(\frac{L_t}{d}\right)^3 \times 10^{-9}$	162	125	97	79
v	1.863	1.863	1.863	1.863
$K \times 10^{13}$	1.35	1.35	1.35	1.35
$\tau\left(\frac{\bar{h}}{\bar{q}_t}\right)^2 \Delta T$	184	119	81	58.5
ΔT	48	32.5	23	16.8
$R_{(OPT)}$	0.00366	0.00373	0.00380	0.00388
ρ	169	169	169	169
$\bar{q}_t$	0.1950	0.2038	0.2080	0.2110
$\frac{w_c}{q_c}$	3.17	3.10	3.09	3.11

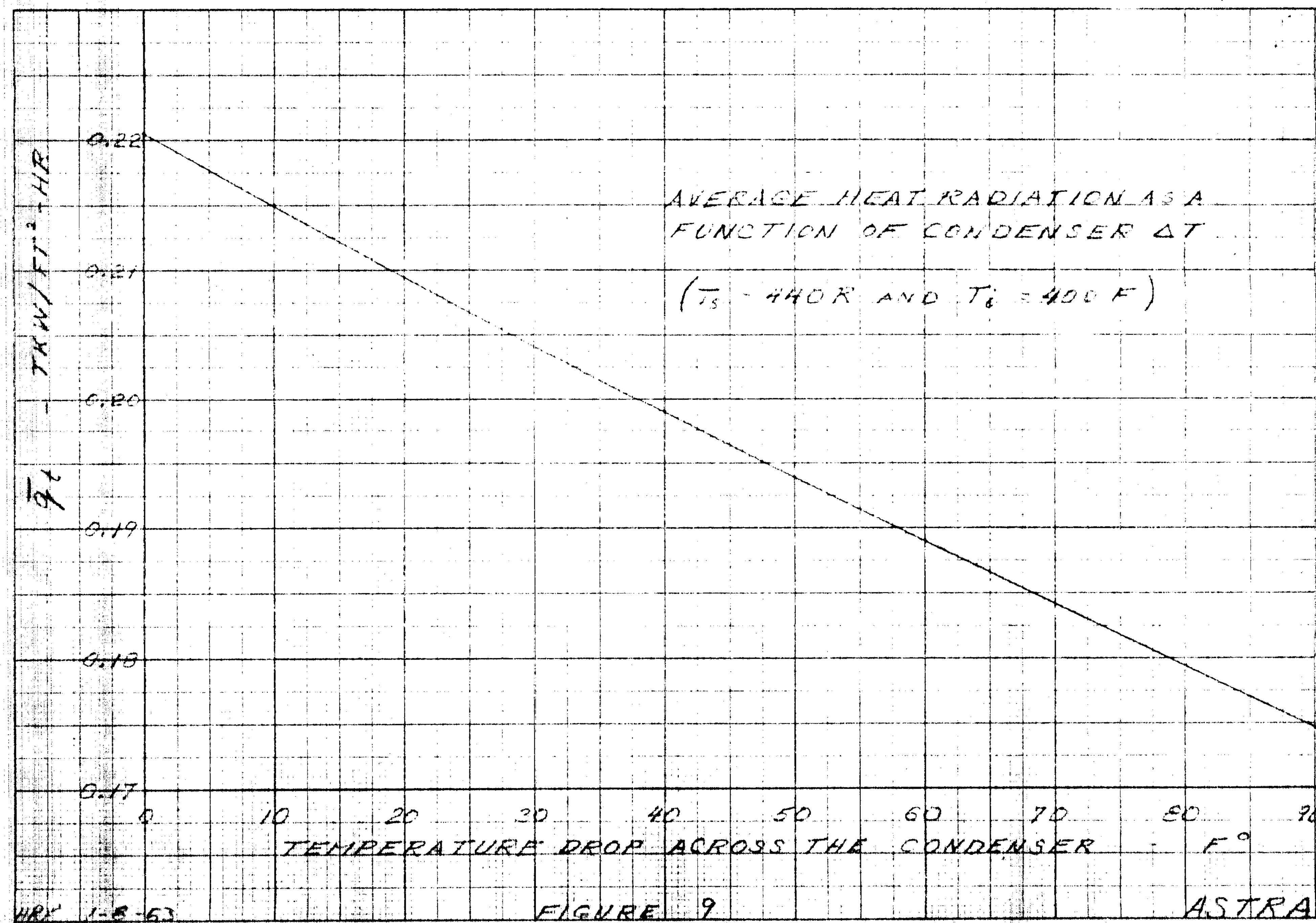

AVERAGE HEAT RADIATION AS A
FUNCTION OF CONDENSER ΔT
($\overline{T_s}$ - 440R AND T_i = 400 F)
$\overline{q}_t$ TKW/FT² - HR
0.22
0.21
0.20
0.19
0.18
0.17
0
10
20
30
40
50
60
70
80
90
TEMPERATURE DROP ACROSS THE CONDENSER - F°
HRK 1-8-63
ASTRA

FIGURE 9

Figure 10 is a plot of the $\frac{w_c}{q_c}$, ΔT relationship for several values of L.

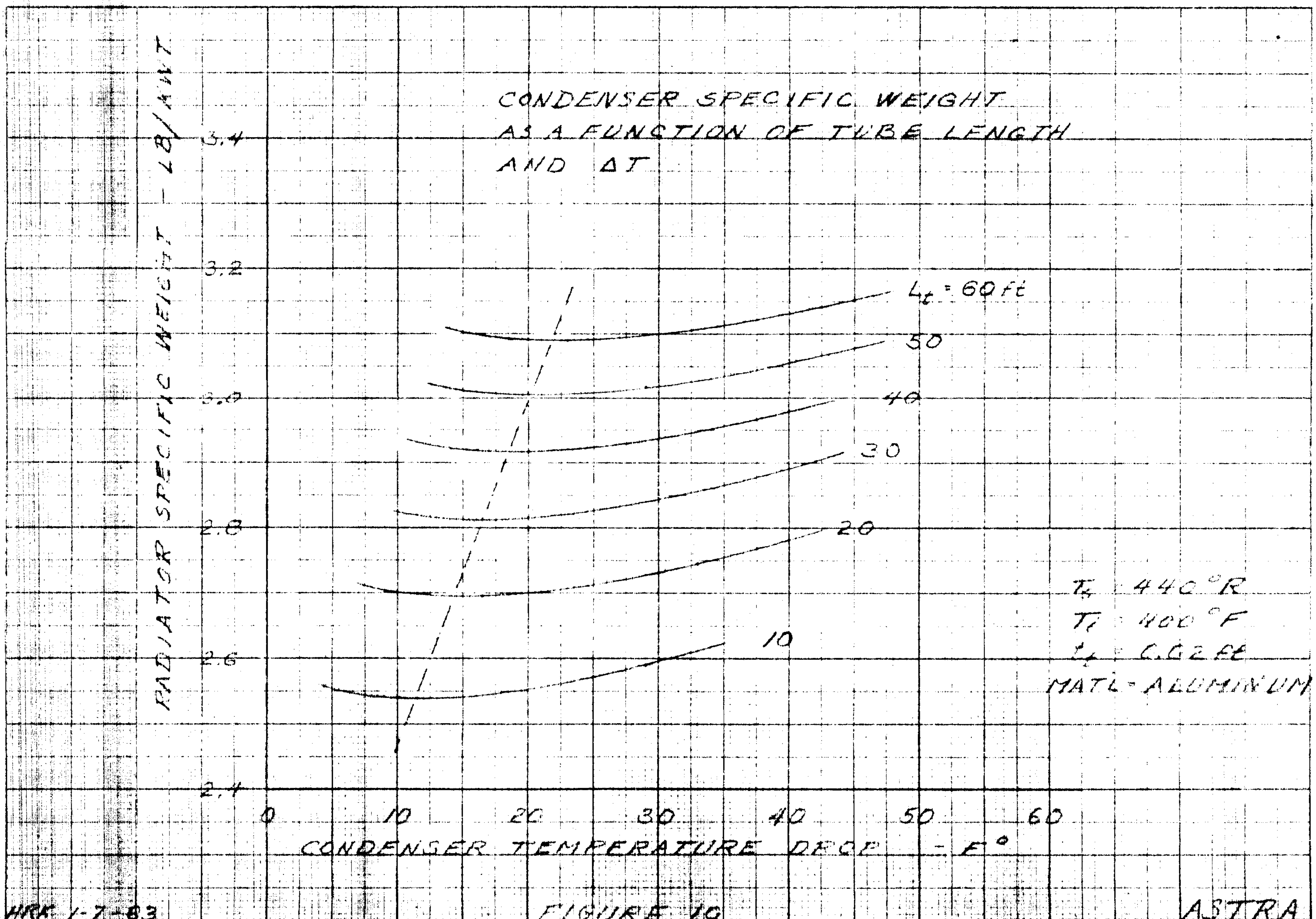

CONDENSER SPECIFIC WEIGHT
AS A FUNCTION OF TUBE LENGTH
AND ΔT
RADIATOR SPECIFIC WEIGHT - LB/KWT
3.4
3.2
3.0
2.8
2.6
2.4
0
10
20
30
40
50
60
CONDENSER TEMPERATURE DROP - F°
L_t = 60 ft
50
40
30
20
10
T_s = 440 °R
T_i = 400 °F
t_t = 0.02 ft
MATL = ALUMINUM
HRK 1-7-63
ASTRA

FIGURE 10

V. THE OPTIMUM HEADER ARRANGEMENT

A 30 ekw plant operating at an average condenser temperature of 390 F will have an overall efficiency of ~ 12%. Thus the heat generated by the reactor would be ~ 250 tkw and the heat rejected by the radiator would be ~ 220 tkw.

The heat radiated from each radiator tube is (from Section III)

$$\frac{Q_c}{N} = L_t \bar{q}_t (C_3 + C_4 B) \; .$$

The number of parallel tubes required is

$$N = \frac{Q_c}{L_t \bar{q}_t (C_3 + C_4 B)} \; .$$

Assuming a symmetric array of two identical radiator "wings", N must be an even number. Also, assuming the turbine is located at the geometrical center of the two "wings," the headers will consist of two pairs of steam and water headers.

For the steam headers it was assumed that the cross sectional area is 4 times the total cross sectional area of the tubes served.

That is:

$$\frac{\pi}{4} d_{sh}^2 = 4 \frac{\pi}{4} d^2 \text{ (tubes served)}.$$

Considering tapered headers the average header area would occur approximately where the average number of tubes was served.

$$\text{Tubes served} = \frac{1}{2}\left(\frac{N}{2} + 2\right) = \frac{N+4}{4}$$

$$\text{And } \frac{\pi}{4} d_{sh}^{\ 2} = 4 \frac{\pi}{4} d^2 \left(\frac{N+4}{4}\right).$$

Then the average inside diameter is <u>approximately</u>

$$\bar{d}_{sh} = d\ (N + 4)^{1/2} .$$

For

$$t_h = 0.02'$$

$$W_{sh} = \frac{\pi}{4}\left(\bar{D}_h^{\ 2} - \bar{d}_{sh}^{\ 2}\right) \rho l_h$$

where,

W_{sh} = weight of one steam header, and

$$W_{sh} = \pi t_h(\bar{d}_{sh} + t_h)\ \rho l_h \quad .$$

For the condensate return headers the water velocity is quite low and the cross sectional area may be assumed to just equal that of the tubes served.

Thus,

$$d_{wh} = \frac{\bar{d}}{2} (N + 4)^{1/2} = \frac{\bar{d}_{sh}}{2}$$

and

$$W_{wh} = \pi t_h \left(\frac{\bar{d}_{sh}}{2} + t_h \right) \rho l_h .$$

The total header weight is:

$$W_{TOTh} = 2 \pi \rho l_h t_h \left(\bar{d}_{sh} + t_h + \frac{\bar{d}_{sh}}{2} + t_h \right)$$

$$= 2 \pi \rho l_h t_h \left(\frac{3}{2} \bar{d}_{sh} + 2t_h \right)$$

$$= 3 \pi \rho l_h t_h \left(\bar{d}_{sh} + \frac{4}{3} t_h \right) .$$

For

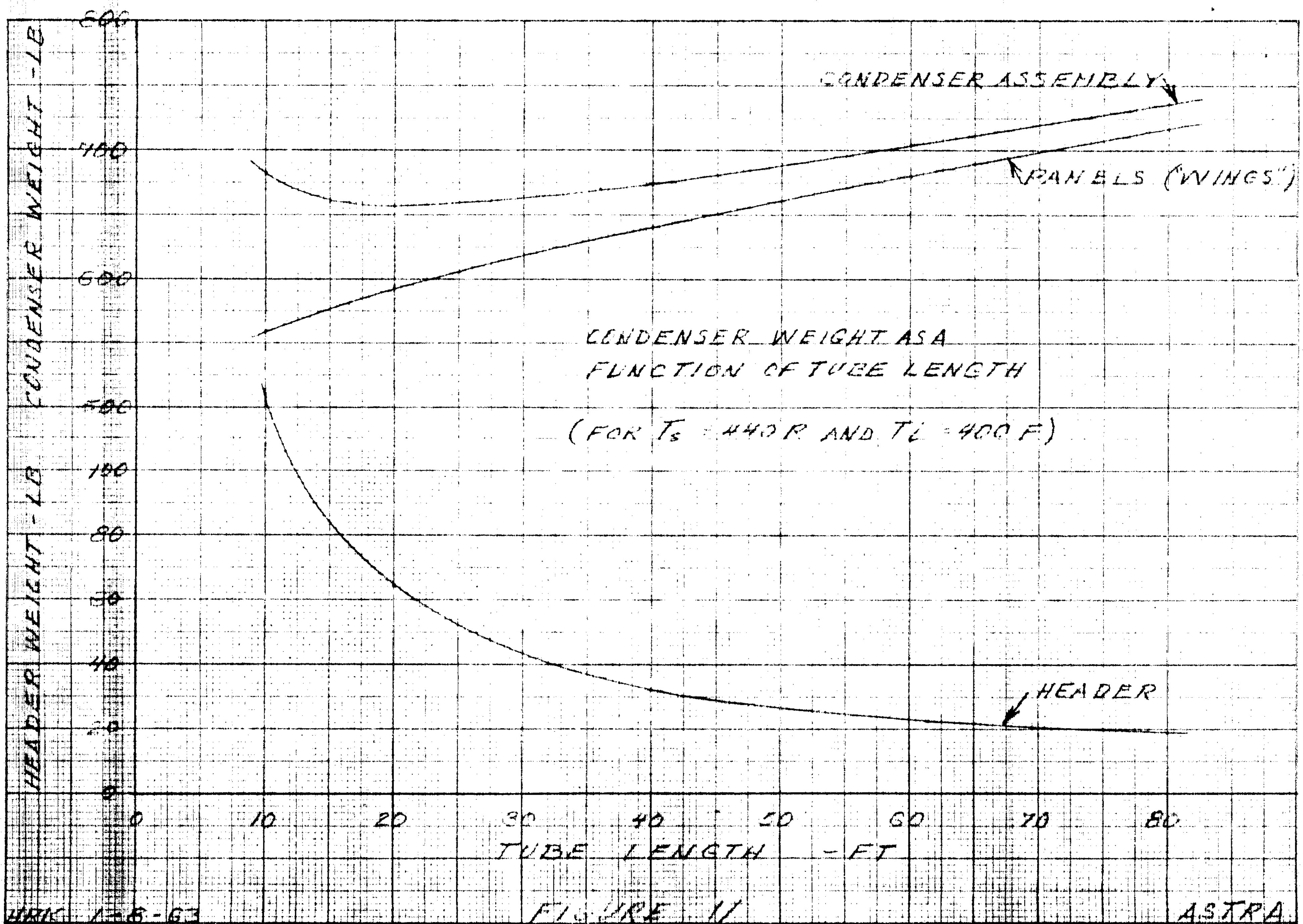
$\rho = 169$ (Al.) and $t_h = 0.02$

$$W_{TOTh} = 31.9 \; l_h \left(\bar{d}_{sh} + 0.0267 \right) .$$

It is assumed that each "wing" of the radiator consists of "folded" tubes attached to headers at the whole radiator centerline. Thus the wing dimension normal to the centerline is approximately $L_t/2$ and the dimension along the centerline is:

$$\frac{N}{2} \times 2 \; (B + D) = N(B + D).$$

By assumption $l_h = \frac{N}{2} (B + D)$.


CONDENSER ASSEMBLY
PANELS ("WINGS")
CONDENSER WEIGHT AS A
FUNCTION OF TUBE LENGTH
(FOR T_s = 440 R AND T_L = 400 F)
HEADER
CONDENSER WEIGHT - LB
HEADER WEIGHT - LB
600
100
80
60
40
20
0
0
10
20
30
40
50
60
70
80
TUBE LENGTH - FT
FIGURE 11
ASTRA

The weight of the tubes and fins alone are just:

$$W_c = \frac{w_c}{q_c} \qquad Q_c = 220 \frac{w_c}{q_c} \quad .$$

Table IV shows the details of the header and weight computation.

Figure 11 shows the tube and fin weights, the header weights, and the weight of the radiator-header assembly all as a function of tube length.

As expected, a minimum radiator weight occurs (for $L_t = 20$). This results in a radiator panel 10 ft wide by 44 ft long. A more reasonable choice structurally is $L_t = 40$ ft. This gives a panel 20 x 22 ft for a weight penalty of about 20 pounds.

Based on a choice of $L_t = 40$ ft, the detailed radiator specifications are given in Section I-C.

TABLE IV. DETERMINATION OF CONDENSER AND HEADER WEIGHTS FOR VARIOUS TUBE LENGTHS

(Q_c = 220 tkw)

L_t	10	20	40	60	80
$\bar{q}_t$	0.2140	0.2122	0.210	0.2084	0.2070
(C_3+C_4B)	0.667	0.690	0.723	0.750	0.776
NL_t	1542	1503	1450	1407	1370
N	154.2	75.2	36.3	23.5	17.1
N*	154	76	36	24	18
d	0.005	0.0072	0.0105	0.0132	0.0157
$(N+4)^{\frac{1}{2}}$	12.57	8.94	6.32	5.29	4.69
$\bar{d}_{sh}$	0.0628	0.0644	0.0663	0.0698	0.0736
$(d_{sh}+0.0267)$	0.0895	0.0911	0.0930	0.0965	0.1003
(B+D)	0.5615	0.580	0.606	0.626	0.644
N(B+D)	86.4	44.0	21.8	15.0	11.6
l_n	43.2	22	10.9	7.5	5.8
W_{TOTh}	123	64	32.3	23.1	18.6
$\frac{W_c}{q_c}$	2.54	2.695	2.917	3.09	3.26
W_c	559	593	641	680	717
W_{c+h}	682	657	673	703	736

VI. APPENDIX

A. METEOROID ARMOR THICKNESS

The expression for the meteoroid armor thickness is (from Loeffler and Lieblein):

$$t = \frac{a}{30.5}\,\gamma\left(\frac{6}{\pi}\right)^{\frac{1}{3}}\left(\rho_p\right)^{\frac{1}{3}}\left(\frac{62.4\rho_p}{\rho_t}\right)^{\phi}\left(\frac{\overline{V}}{c}\right)^{\theta}\left(\frac{0.0417\alpha AH}{-\ln P(0)}\right)^{\frac{1}{3\beta}}\left(\frac{2}{3n\theta\beta + 2}\right)^{\frac{1}{3\beta}}$$

where t = required tube wall thickness in ft.

a = spall adjustment = 1.75

$\gamma = 2$

ρ_p = meteoroid particle density = 0.6 g/cm^3

ρ_t = target material density in lb/ft^3 (for aluminum $\rho_t = 169$)

$\phi = \frac{1}{2}$

$\overline{V}$ = meteoroid velocity = 98,400 ft/sec

$c = 12\left(\frac{E_t g}{\rho_t}\right)^{1/2}$ where E_t = Young's modulus

(for aluminum $c = 12\left(\frac{9.3 \times 10^6 \times 32.2}{169}\right)^{1/2} = 15{,}960$

$\theta = \frac{2}{3}$

$\alpha = 1.01 \times 10^{-10}$

$\beta = 1.34$

A = Vulnerable or exposed area - ft^2 ($= \Sigma\pi DL$), where D is the outside diameter of the tube or headers, and L is the length of tube or headers

H = mission time in hours = 10,000

P(0) = design probability for radiator survival

n = 1.

(The above values for ρ_p, α and β are mid-December 1962 numbers obtained from Mr. Ream of LRC.)

Inserting the above numbers and performing the indicated operation:

$$\frac{a\gamma}{30.5} = \frac{1.75 \times 2}{30.5} = 0.1147$$

$$\left(\frac{6}{\pi}\right)^{1/3} = 1.911^{1/3} = 1.241$$

$$(\rho_p)^{-1/3} = \frac{1}{0.6^{1/3}} = \frac{1}{0.8434} = 1.184$$

$$\left(\frac{62.4\rho_p}{\rho_t}\right)^{\Phi} = \left(\frac{62.4 \times 0.6}{169}\right)^{1/2} = 0.2214^{1/2} = 0.471$$

$$\left(\frac{\overline{V}}{c}\right)^{\theta} = \left(\frac{98,400}{15,960}\right)^{2/3} = (6.17)^{2/3} = 3.36$$

$$\left(\frac{0.0417\alpha AH}{-\ln P(0)}\right)^{\frac{1}{3\beta}} = \left(\frac{4.21 \times 10^{-8} A}{-\ln P(0)}\right)^{0.249}$$

For $P(0) = 0.9$, $-\ln P(0) = 0.1054$.

Assume $A = 180$ ft^2,

then

$$\left(\frac{4.21 \times 10^{-8} \times 180}{0.1054}\right)^{0.249} = (0.719 \times 10^{-4})^{0.249}$$

$$= 0.0921$$

$$\left(\frac{2}{3n\theta\beta + 2}\right)^{\frac{1}{3\beta}} = \left(\frac{2}{3(\frac{2}{3})\ 1.34 + 2}\right)^{0.249} = (0.427)^{0.249}$$

$$= 0.809$$

and

$$t = 0.1147 \times 1.241 \times 1.184 \times 0.471 \times 3.36 \times 0.0921 \times 0.809$$

$$t = 0.0199 \text{ ft., say, } 0.02 \text{ ft.}$$

The armor thickness for any vulnerable area can be obtained from

$$t = t_o \left(\frac{A}{A_o}\right)^{0.249} = 0.02\left(\frac{A}{180}\right)^{0.249}.$$

Similarly, the armor thickness for any assumed survival probability can be obtained from

$$t = t_o \left(\frac{-\ln \, (P(0))_o}{-\ln \, P(0)} \right)^{0.249} = 0.02 \left(\frac{0.1054}{-\ln \, P(0)} \right)^{0.249} .$$

Figure 12 shows the armor thickness required for various values of vulnerable area and survival probability.

The terms "armor thickness" and "tube wall thickness" are used interchangeably since the required protective armor thickness provides many times the cross-sectional wall area necessary to contain the hoop stresses.

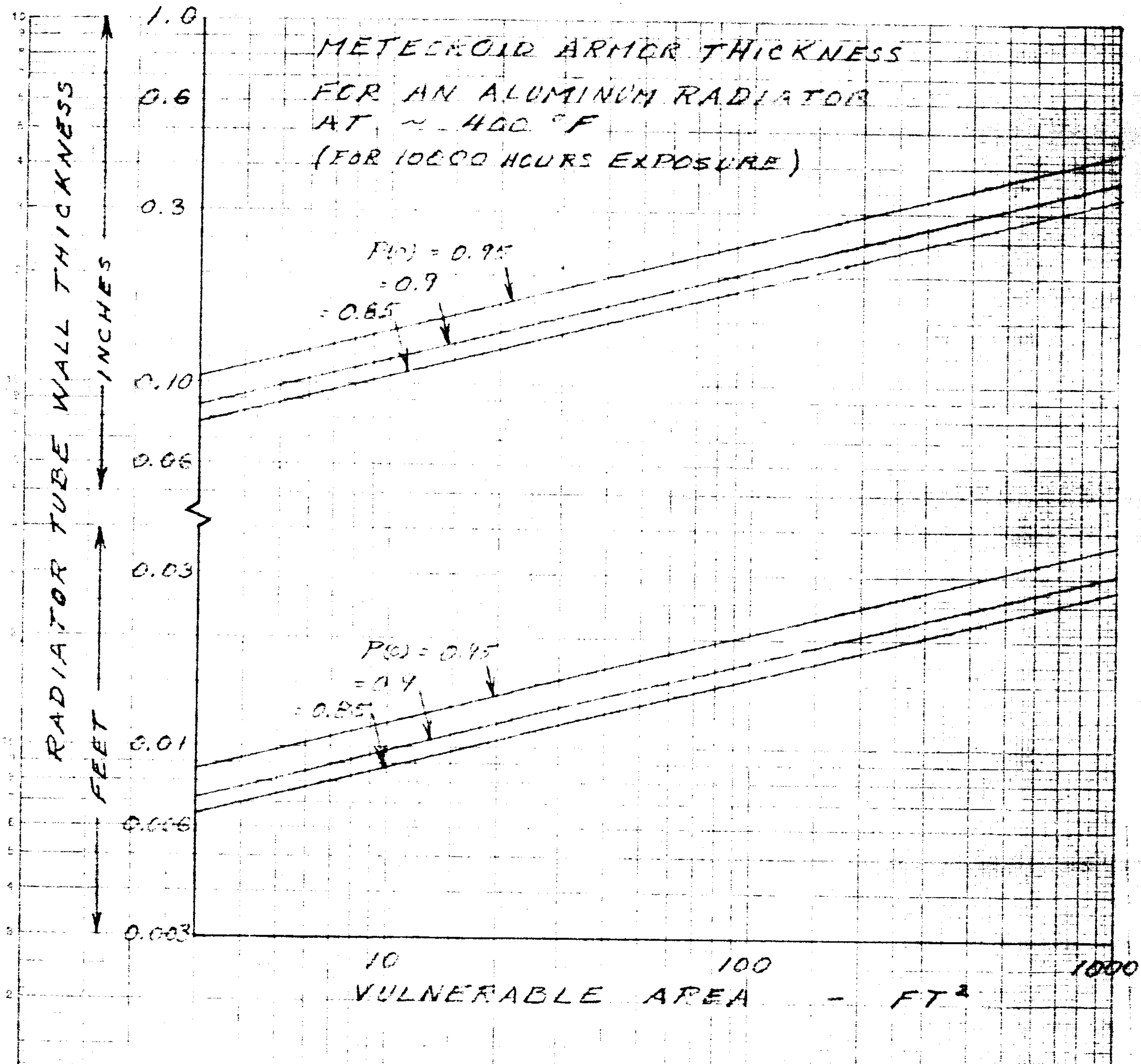

METEOROID ARMOR THICKNESS
FOR AN ALUMINUM RADIATOR
AT ~ 400 °F
(FOR 10000 HOURS EXPOSURE)
P(0) = 0.95
= 0.9
= 0.85
P(0) = 0.95
= 0.9
= 0.85
RADIATOR TUBE WALL THICKNESS
INCHES
FEET
1.0
0.6
0.3
0.10
0.06
0.03
0.01
0.006
0.003
10
100
1000
VULNERABLE AREA - FT²
HRK 1-3-63
ASTRA

FIGURE 12

B. PRESSURE DROP ACROSS SPACE RADIATORS, IN WHICH STEAM IS CONDENSING

Accurate calculation of the pressure drop, which occurs as the working fluid condenses in a space radiator is an important factor in the design optimization of a thermal power system. The pressure drop determines, to a large extent, the effective condensing temperature, which is also (very nearly) the radiating temperature. The relationship between saturation pressure drop and saturation temperature drop varies over a large range for different flow media. The condensing temperature of steam is less effected by changes in pressure, however, than other working fluids, such as mercury, sodium, and potassium.

A literature search was made of previous pressure drop investigations. It was found that they all followed similar lines of reasoning. Many of the techniques were quite complex and attempted to be most scholarly; all too frequently, however, these "pure" techniques led to tedious or unsolvable mathematical equations, and simplifying assumptions had to be made.

Basically, pressure drop is related to two independent effects:

1. Drag friction between tube walls, and the moving fluid, and,

2. Conversion of velocity pressure to static pressure caused by the decrease in momentum of the vapor as it condenses.

It should be noted that in a condensing situation, these effects are of opposite sign; i.e., the first effect causes a pressure loss while the second tends to cause a pressure rise. (Optimum radiator weight design will always require a net pressure drop.)

In calculating the pressure drop $(\Delta P)_{TP}$ due to friction, it is necessary to consider the dynamics of two-phase flow. One of the more generally accepted solutions to problems of two-phase flow through tubes was published by Lockhart and Martinelli(1) and Martinelli, et al.(2) The correlation proposed by Lockhart and Martinelli is based on the hypothesis that the pressure drop for simultaneous flow of two phases is equal to the pressure drop which would occur if one of the phases were flowing alone, multiplied by a correction factor; this factor depends on the state of the flow regime.

Several modifications and extensions have been attempted[3-6] to improve the usefulness of the Lockhart and Martinelli correlations; they have, however, still retained the major disadvantage of having to know the particular flow regime before deciding on the appropriate correlation chart. In an attempt to overcome this disadvantage, Bergelin, Gazeley presented a correlation for predicting gas-phase pressure drop for stratified and annular flow patterns.[7-9] Treatment of each of the two phases as a pseudo single-phase fluid has been suggested and applied with some success to flow-rate-pressure-drop calculations by several investigators.[10,11] It seems, however, that "some success" may include deviations as high as 30% between calculated and experimentally observed two-phase pressure drops.[1,2,12]

The pressure change in the second effect $(\Delta P)_M$, caused by the change in momentum, is analyzed by principles of simple gas dynamics.[13] The cited reference is typical of derivations for $(\Delta P)_M$ and necessarily makes assumptions which incur some error.

All derivations studied obtain the net pressure change across the condenser (ΔP) by summing the two effects:

$$\Delta P = (\Delta P)_F + (\Delta P)_M \quad .$$

This is the necessary method of analysis for liquid-metal systems because the large density changes from vapor to liquid place considerably emphasis on the $(\Delta P)_M$ term. However, when steam-water is used as the working fluid, the method of pressure drop analysis can be greatly simplified. This is because for optimum pressure drop (ΔP) the $(\Delta P)_M$ is subordinated manyfold by the $(\Delta P)_F$ term. In accordance with this, Kern(14) has suggested a means for computing the pressure drop across steam condensers:

> "In the condensation of a pure saturated vapor, the vapor enters the condenser at its saturation temperature and leaves as a liquid. The pressure drop is obviously less than that which would be calculated for a gas at the inlet specific gravity of the vapor and greater than that which would be computed using the outlet specific gravity of the condensate. The mass velocity of inlet vapor and outlet liquid are, however, the same. In the absence of extensive correlations reasonably good results are obtained when the pressure drop is calculated for a mass velocity using the total weight flow and the average specific gravity between inlet and outlet. This method is further simplified in the condensation of steam, by taking one-half the conventional pressure drop computed entirely on inlet conditions."

The resulting expression is on the conservative side, since the mass velocity of the vapor decreases nearly linearly from inlet to outlet, whereas the pressure drop decreases as the square of the velocity.

Thus, using Kern's method, an expression is developed for pressure drop of condensing steam in a tube:

$$\Delta P_{SP} = \frac{1}{2g} \rho_v V^2 \left(f_w \frac{L_t}{d}\right) \tag{15}$$

where ΔP_{SP} is the pressure drop for single phase flow of vapor through the tube - lb/ft^2

g is the gravity constant (in this case 4.18×10^8 ft/hr^2)

ρ_v is the density of the vapor in lb/ft^3

V is the flow velocity in ft/hr

f_w is Weisbach's friction factor

L_t is the tube length in ft.

and d is the tube diameter in ft.

But

$$V^2 = \left(\frac{4}{\pi}\right)^2 \frac{m^2 v^2}{d^4}$$

where m is the mass flow rate in lb/hr

and v is the specific volume of the vapor in ft^3/lb.

Thus

$$\Delta P_{SP} = \frac{1}{2g} \quad \rho_v \left(\frac{4}{\pi} \right)^2 \frac{m^2 v^2}{d^4} \quad \left(f_w \frac{L_t}{d} \right) .$$

Assuming f_w is constant at 0.02[16]

$$\Delta P_{SP} = \frac{f_w}{2g} \left(\frac{4}{\pi} \right)^2 \frac{m^2}{d^4} v \left(\frac{L_t}{d} \right)$$

$$= 2.88 \times 10^{-11} \frac{m^2}{d^4} \quad v \quad \left(\frac{L_t}{d} \right) .$$

Assuming Kern's suggestions and dividing by 144

$$\Delta P = \frac{\Delta P_{SP}}{2 \times 144}$$

where ΔP is in lb/in^2

$$\Delta P = 1.35 \times 10^{-13} \frac{m^2}{d^4} \quad v \quad \left(\frac{L_t}{d} \right) .$$

The above expression is subject to check by comparing with test results.

Included in a report by TAPCO,[17] are experimental results obtained for condensing steam in tubes with a uniform heat transfer rate.

Experimental Data:

Inlet Conditions:

P - 16.2 psia

$t = 240^{o}F$

m = 1.23 lb/hr/tube (mass flow rate)

$L_t = 4$ ft (tube length)

d = 0.0121 ft (tube I.D.)

By Kern's Method:

$$\Delta P = \frac{1.35 \times 10^{-13} \; m^2 v}{d^4} \left(\frac{L_t}{d}\right)$$

$$v \Big|_{\text{Inlet conditions}} = 25.4 \text{ ft}^3/\text{lb}$$

$$\Delta P = \frac{1.35 \times 10^{-13} \; (1.23)^2 \; (25.4)(4)}{(.0121)^5}$$

$\Delta P = 0.08$ psi.

Observed results (vertical tube)

ΔP (across cond.) = 150.7 psf (Vertical tubes)

ΔP (gravity head) = 137.5 psf

ΔP (net) = 13.2 psf pressure drop across condenser with gravity effect removed.

COMPARISON:

Pressure drop observed 13.2 psf = 0.0916 psi

Pressure drop calculated = 0.08 psi

Percent deviation: (-)* 13%

This deviation was expected as there were 23 degrees of superheat at the inlet of the test tube. Condensing didn't begin until vapor was some distance down tube. Thus a portion of the tube operated under single phase flow conditions and at maximum velocity - hence maximum pressure drop.

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* Minus sign indicates the calculated value less than the observed value.

REFERENCES

1. R. W. Lockhard and R. C. Martinelli, "Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes," Chem. Eng. Progr., Vol. 45, 1949, p. 39-48.

2. R. C. Martinelli, L. M. K. Boelter, T. M. Taylor, E. G. Thomsen, and E. H. Morin, "Isothermal Pressure Drop for Two-Phase, Two-Component Flow in a Horizontal Pipe," Trans. ASME, February, 1944, p. 139-151.

3. O. Baker, "Simultaneous Flow of Oil and Gas," Oil Gas J., Vol. 53, No. 12, 1954, p. 185-195.

4. J. M. Chenoweth and M. W. Martin, "A Pressure Drop Correlation for Turbulent Two-Phase Flow of Gas Liquid Mixtures in Horizontal Pipes," Petrol. Refiner, Vol. 34, 1955, p. 151.

5. R. C. Reid, A. B. Reynolds, A. J. Diglio, I. Spiewak, and D. H. Klipstein, "Two-Phase Pressure Drops in Large Diameter Pipes," AIChE J., Vol. 3, No. 3, 1957, p. 321-324.

6. G. E. Alves, "Co-current Liquid-Gas Flow in a Pipeline Contractor," Chem. Eng. Progr., Vol. 50, 1954, p. 449-456.

7. O. P. Bergelin and C. Gazely, "Co-current Gas-Liquid Flow. I. Flow in Horizontal Tubes," Heat Transfer and Fluid Mechanics Institute, Berkeley, California, 1949.

9. O. P. Bergelin, "Flow of Gas Liquid Mixtures," Chem. Eng., Vol. 56, 1949, p. 104.

10. A. F. Bertuzzi, M. R. Tek, and F. H. Poettmann, "Simultaneous Flow of Liquid and Gas Through Horizontal Pipes," Trans. AIME, Vol. 207, 1956, p. 17.

11. M. W. Benjamin and J. G. Miller, "Flow of a Flashing Mixture of Water and Steam through Pipes," Trans. ASME, Vol. 64, 1942, p. 657.

12. Electro-Optical Systems, Inc., Report 310-Final, 31 Oct. 60, Sect. C-7.

13. N. W. Snyder, "Two-Phase Flow," Space Power Systems, Academic Press, New York, 1961, p. 305.

14. Donald Q. Kern, Process Heat Transfer, McGraw-Hill Book Co., Inc., New York, 1950, p. 273.

15. Ralph G. Hudson, Hudson's Engineering Manual, John Wiley & Sons, New York, 1953, p. 151.

16. L. S. Marks, Mechanical Engineer's Handbook, McGraw-Hill Book Co., New York, 1951, 5th Ed., p. 250.

17. "Space Radiator Study," TAPCO ASD-TR 61-697, 30 April 62, p. 171.